

DEVELOPMENT OF REPAIR KITS FOR RECIPROCATING/ROTARY COUPLINGS OF TRANSPORTATION AND PRODUCTION EQUIPMENT

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ABSTRACT

Sliding and rolling bearings are applied in transportation and production equipment, these bearings operate under high loads in reciprocating/rotary mode. They are used in railroad, road, and agricultural transport for suspensions, dampers, steering, shaft drives, in garment equipment, mining, oil and gas production and processing, as well as some other industries. However, their operation lifetime is insufficient and recovery of such couplings by conventional methods comprised of replacement with repair kits of similar design does not solve this problem.

As assumed in this article, the operation lifetime of couplings operating under heavy loading in reciprocating/rotary mode can be increased by improved recovery of their operability using repair kits based on new operation principles of tribocouplings, that is, sliding bearings with movable spring bearing boxes. The design of repair kit design of rubber bushing in the form of sliding bearing with movable conical spring boxes has been theoretically substantiated. Mathematical models of size variations of movable conical spring box of sliding bearing of damper rubber bushing have been developed.

Theoretical importance of the work is comprised of the expanded proven concepts of recovery of damper rubber bushings using repair kits with movable spring boxes. Practical significance is that the developed repair kits of damper rubber bushing are characterized by operation lifetime higher by 1.5...2 times in comparison with regular kits. The obtained results can be applied for development of new engineering solutions, for improvement of recovery of transportation and production equipment with couplings operating under high loads in reciprocating/rotary mode.

KEYWORDS: Repair Kit, Spring Boxes, Vehicle Suspension & Dampers

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1. INTRODUCTION

Operation of transport and technological equipment, the technical condition of their systems and assemblies changed, the main reasons for which are various types of wear, plastic deformation, fatigue and temperature failure, fretting corrosion [1-5]. This can lead to a complete or partial loss of operability of the unit, assembly or part, that

is, to its failure or malfunction, the elimination of which is carried out through preventive and repair work, as well as due to the restoration of their operability with a simultaneous increase in operational durability.

Transportation and production equipment operates with sliding and rolling bearings under high loads in reciprocating/rotary mode. Such bearings are used in railroad, road and agricultural transport in suspensions, dampers, steering, shaft drives, in garment equipment, mining, oil and gas production and processing, as well as some other industries. Manufacturing companies of the equipment are requested to increase operation lifetime retaining reliability of parts, units, assemblies and systems. However, at the design and manufacturing stages, these problems are solved insufficiently, and during operation it is required to carry out functional tuning in order to improve operation quality [6]. Functional tuning, besides all, assumes application of repair kits in units and assemblies in order to recover their working state and even for improvement of their operation lifetime.

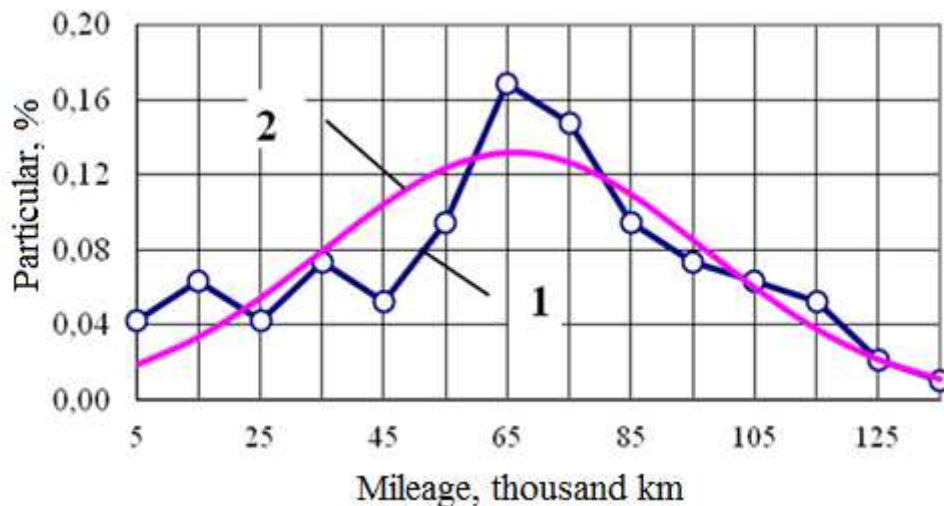
Numerous works are devoted to provision of operability of transportation equipment in the course of operation lifetime: Avdon'kin, Apsin, Grebennikov, Grigor'ev, Gurvich, Denisov, Dyumin, Zvyagin, Kanarchuk, Kramarenko, Kuznetsov, Novikov, Savel'ev, Sheinin, N. Gkikas, J. Little, D. Cormick, S. Bennett, I. A. Norman, and others [7, 9, 10, 28, 29, 30].

The analysis of the operational reliability of cars indicated that the elements of the front and rear suspensions do not have the same durability within the operational period [8]. Thus, a significant proportion of failures of its elements (about 90%) occurred in telescopic racks, upper and ball bearings, rear shock absorbers as well as wheel hub bearings (table 1).

Table 1: Distribution of Failure Recurrence of the Main Suspension Elements and the Complexity of their Elimination (for VAZ Family Cars in Samara City)

| Name of Element | Frequent Refusal, % | Unit Quantity of Work, % |
|---------------------|---------------------|--------------------------|
| Telescopic stand | 25,8 | 30,1 |
| Ball joint | 21,6 | 19,7 |
| Rear shock absorber | 17,2 | 11,7 |
| Front wheel bearing | 15,5 | 21,6 |
| Upper cradle | 5,9 | 7,1 |
| Bearing of backhub | 3,2 | 2,5 |
| Stretching bracket | 2,4 | 0,7 |
| Front hub | 2,2 | 2,1 |
| Rear hub | 1,0 | 0,4 |
| Stabilizer stand | 1,0 | 0,1 |
| Stretching | 0,7 | 0,2 |
| Springs | 0,6 | 0,8 |
| Swivel fist | 0,5 | 0,7 |
| Wishbone | 0,3 | 0,3 |
| Other | 2,7 | 2,0 |

The share of telescopic racks and rear shock absorbers accounts for 25.8% and 17.2% of refusal (Table 1). Service stations statistics, the life of standard shock absorbers depending on operation of the car, the driving style of its owner is no more than 70 thousand km (Figure 1).



**Figure 1: Time to Failure of Rear Shock Absorbers of VAZ Family Cars in Samara
(1 and 2 - Actual and Estimated Distribution).**

The defect “breakage of the eye” is characterized for the rear shock absorbers, which was for 10-15% of cases. In addition, it is possible for a rupture of the silent block of the rear shock absorber.

Silent blocks of shock absorbers and cardan joints are one of the most common and characteristic mates of transport equipment operating under heavy loads in a rotary mode.

The loss of performance of silent blocks due to wear and fatigue damage occurs as a result of stresses in the rubber that occur when the load on the car changes, as well as a result of multidirectional forces that occur on rough roads.

In the oscillatory cardan joint with small amplitudes and high normal loads, dents from needles are formed on the working surfaces of the cup and spike of the cross, called “false brinelling”.

Such problems researched: V. B. Gurik, B. D. Voronkov, V. B. Cherkunov, Yu. P. Busarov, B. V. Cherkunov and A. E. Tatatchenko, M. P. Kopak and N. P. Kopak V. G. Kuranov, A. N. Vinogradov [11,12,13].

The problem of reliability and wear resistance of bearings for the reciprocating (oscillatory) mode was not solved [14]. The restoration of such mates by traditional methods by replacing with repair kits similar in design is not effective, since the durability remains unsatisfactory.

Therefore, it is important to search for new engineering approaches aiming at improvement of lifetime of couplings of transportation and production equipment operating under high loads in reciprocating/rotary mode, as well as at improvement of their operability recovery using repair kits.

This work substantiates theoretically the design of repair kit of rubber bushing containing tribocoupling based on new operational principle: sliding bearing with movable conical spring boxes.

2. METHODS

Operation lifetime of tribocouplings, in particular: sliding bearings operating under high loads in reciprocating/rotary mode, can be improved by using concepts of Prof. Zhukovsky about motion without friction leading to its decrease in working body [15].

The first concept was comprised of friction compensation by auxiliary counter-motion of intermediate support driven by external energy source. As we see at Figure 2, when the even and odd threads move in different directions, two opposite friction forces arise:

$$F_1 = fG_1 \text{ and } F_2 = fG_2,$$

where G_1 and G_2 are load parts of the located on even and odd threads;

f - the coefficient of friction.

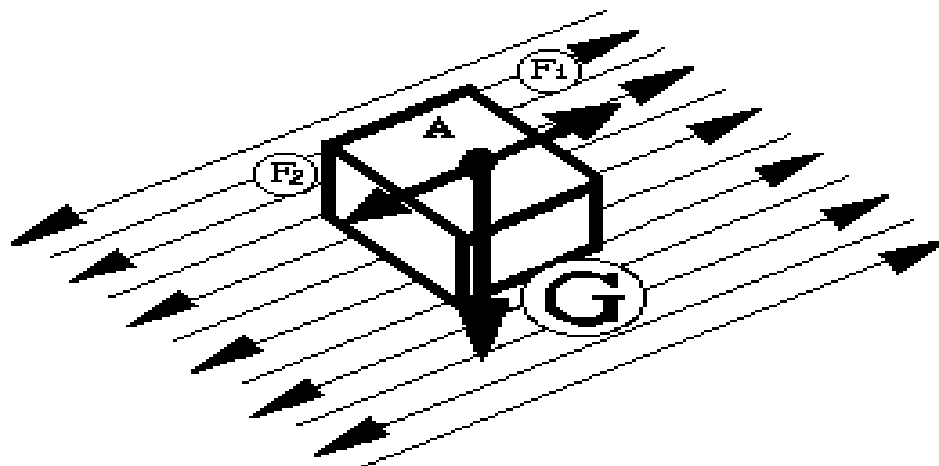


Figure 2: N. E. Zhukovskiy Idea of Friction Compensation in the Oncoming Movement of an Intermediate Engine Mounts.

If the system is symmetrical and $G_1 = G_2$, then $F_1 = F_2$, and the load G will move in the direction of movement of the threads without friction (for those cases when the friction force is independent of speed).

Professor N. E. Zhukovskiy's idea was confirmed with the help of a pendulum, which was suspended on a sliding engine mount with an insert in the form of a bronze bushing and made only a few oscillations until it stopped. But, after the insert sleeve was cut in half (Figure 3) and each half was forced to rotate in the opposite direction from an external source, this pendulum made several thousand oscillations to a stopping.

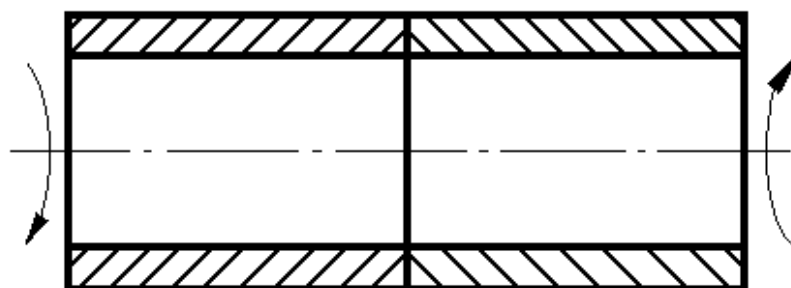


Figure 3: The Cut Sleeve of the Pendulum.

The second idea is illustrated in Figure 4, and it is used (instead of oncoming) of the lateral auxiliary movement of the platform, the speed of which should be much greater than the sliding speed of the G load on A platform.

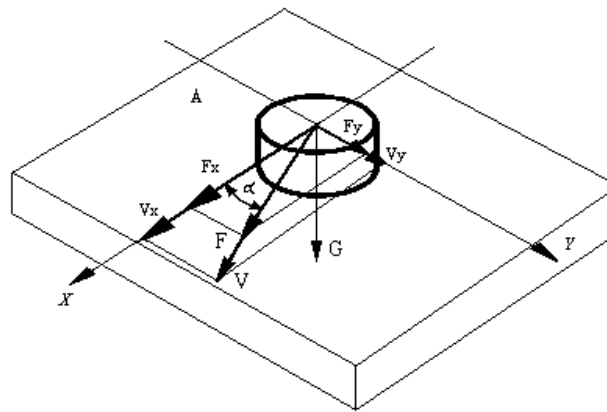


Figure 4: Illustration of the Second Idea of Friction Compensation using Lateral Auxiliary Platform Movement.

As we see at Figure 4, at $V_x \gg V_y$ the friction force in the Y direction will be less than in X, which depends on the angle α . It is assumed that the components of the friction force will also be located along the X and Y axes [14].

$$F_y = f \cdot G \cdot \sin \alpha = f \cdot G \cdot \frac{V_y}{\sqrt{V_y^2 + V_x^2}}.$$

$$F_y = f \cdot G \cdot \sin \alpha = f \cdot G \cdot \sin \alpha \rightarrow 0 \text{ при } V_x \gg V_y \rightarrow \sin \alpha \rightarrow 0 \text{ и } F_y \rightarrow \min.$$

To implement the second idea, various designs can be used, including without an external energy source, for example, due to the residual unbalance of the rotating mass (Figure 5).

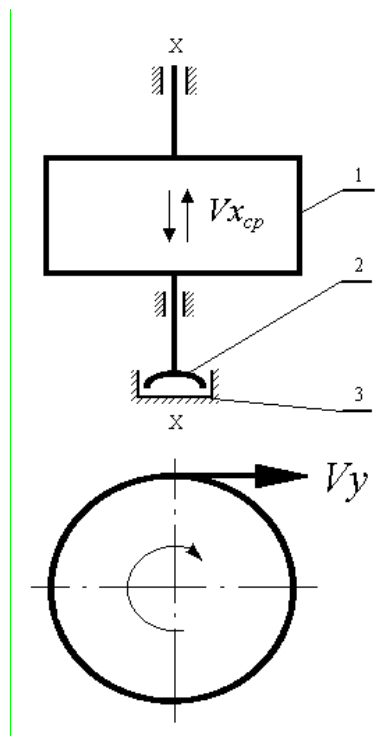


Figure 5: Illustration of Friction Reduction due to Increased Vibration along the X Axis: 1 - Rotating Mass; 2 - Elastic Engine Mount; 3 - Thrust Bearing.

With an increase in the frequency and amplitude of the vibration, which can be increased by introducing the

elastic element 2, along the X axis, the average speed of the auxiliary motion $V_{x_{cp}}$ can be much higher than the rotation speed V_y and, thus, the friction force during rotation relative to the X axis is reduced.

These ideas were most developed in gyroscopic devices, for which special “reversible-rotating bearings” were created — ball bearings with two rows of balls and intermediate rings that rotated in opposite directions through the gear transmission [16] (Figure 6).

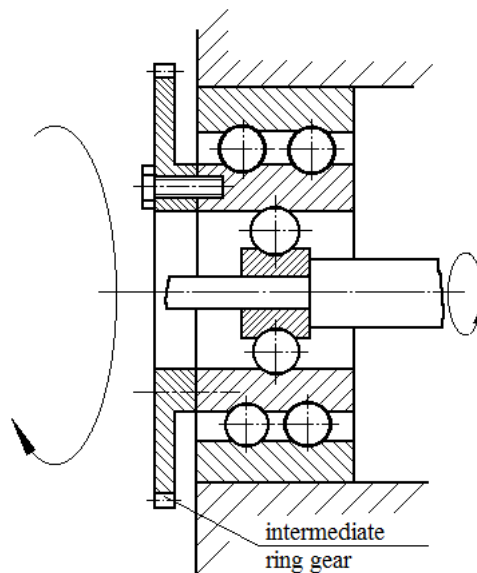


Figure 6: Forced Rotation of the Intermediate Ring of the Gyro Support from an Additional Electric Motor.

The implementation of these ideas with the help of forced rotation or vibration of the intermediate support confirmed the possibility of reducing friction in the working body. In this case, a decrease in the adhesion component of friction is observed during the transition from the state of rest to motion with increasing speed. The transition from rest to motion and discreteness of friction are due to the presence of elastic displacement and the formation of adhesive bonds. When the stiffness decreases in the elastic element and the sliding speed, the rest time increases, a greater number of frictional bonds have time to form, which increases the rest friction force up to the occurrence of setting. This is well observed when there is an elastic link in the drive (in moving electrical contacts [17], distribution mechanisms of automobile engines, refrigeration compressors, etc.).

In internal combustion engines, the introduction of an elastic link into the drive of the distribution mechanism (chain, toothed belt) led to the emergence of the setting and abnormal wear of the cam of the camshafts (Figure 7). This was also affected by other factors: high specific loads on the top of the cam, tearing the oil film (in the oil starvation mode), insufficient hardness of the contacting materials, the desire of drivers to regulate the stability of idling at minimum speed, etc.

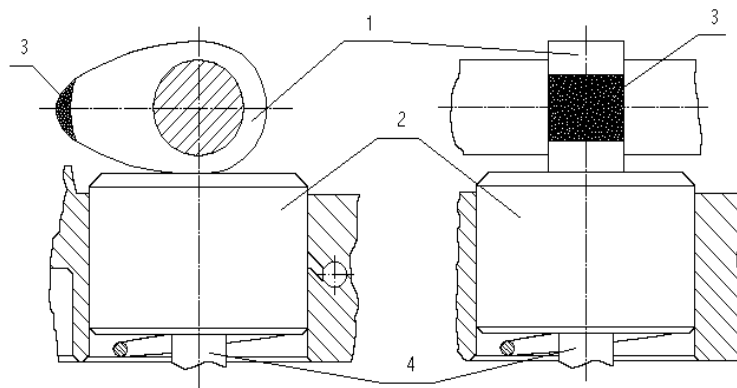


Figure 7: Abnormal Wear of the Cams in Internal Combustion Engines with an Elastic Link of the Drive of the Distributing Mechanism and Hydraulic Gap Compensation: 1 - Camshaft cam; 2 - Hydraulic Compensator; 3 - Cam Wear; 4 - Valve Stem.

However, in "low-speed" engines, where a hard drive was used through gear gears, abnormal cam wear was not observed for many years of operation.

In engines with a lower camshaft and transmitting movement to the valves through pushers and rods (Figure 8), the absence of abnormal cam wear can be explained by the actual implementation of professor N. E. Zhukovskiy second idea - the presence of auxiliary lateral motion (rotation of the pusher) and the translation of sliding friction into rolling friction, which is achieved by displacing the cam relative to the axis of rotation of the pusher.

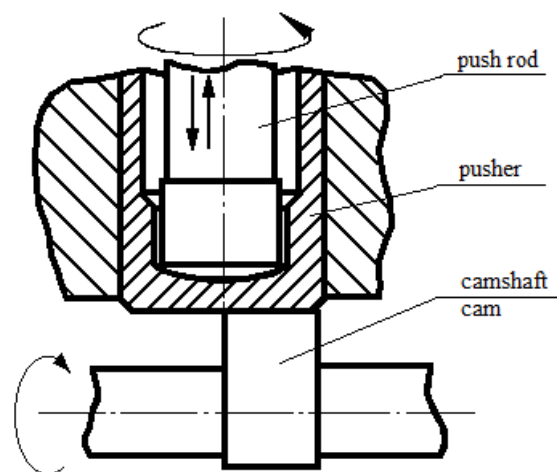


Figure 8: Scheme of the Implementation of Auxiliary (Lateral) Movement in the Support due to the Displacement of the Axis of Rotation of the Cam (without using External Energy).

A feature of the scheme is that it does not require external energy to create an auxiliary motion of the support. The pusher rotation is carried out by friction, but there is no increase in the total energy consumption. The working surfaces of the cams after running in become mirror, operate in a mode of stable normal oxidative friction during long-term operation and have no noticeable wear.

Professor N. E. Zhukovskiy idea on motion without friction" (rotation of an intermediate support) without using an external energy source for this is partially realized in a sliding bearing for reciprocating motion, developed by scientists of SSTU named after Yu. A. Gagarin (Figure 9) [18, 19].

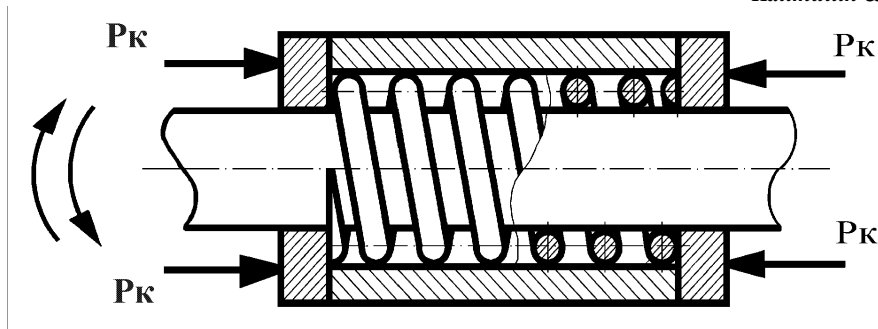


Figure 9: Scheme of a Bearing with a Movable Spring Sleeve.

The principal novelty of the bearing design is that it has a movable liner in the form of a cylindrical spring (an intermediate element), which in the oscillatory mode is forced to rotate only one way, and thus uniform wear and lubricant distribution are achieved by reducing the adhesion component of friction and the appearance of the “effect ratchet” [20]. This bearing has tribological principles - the conditions for the activation of the working surface by plastic deformation and the suppression of oxidative processes. The activation of the working surfaces by plastic deformation is carried out by installing an elastic spring liner between the outer and inner bushings so that there is a slight interference on the working surfaces of the liner. When the bearing operates (when turning in one direction), the interference increases on one of the working surfaces, and decreases on the other until a clearance and slippage form. When turning in the opposite direction on that surface where there was a gap, interference will occur, and vice versa. Suppression (limitation) of oxidative processes on the working surfaces of bearings is ensured constructively, i.e. gland seals are installed to prevent oxygen and other oxidizing agents from accessing work surfaces or technologically by introducing inhibitors into the lubricant.

The bearing can be used instead of the needle bearings of the driveshaft, silent blocks of the suspension, steering hinges and other hinged nodes operating in the rotary mode.

Researchers showed that spring liners, in comparison with needle roller bearings, are devoid of their main drawback - dents from needles do not form on working surfaces (the phenomenon of “false brinelling”). However, the cylindrical spring liner revealed drawbacks due to the need to ensure the required fit - the use of high-precision equipment and expensive tools in the manufacture and it is difficult to assemble using selective method.

To eliminate the shortcomings of the cylindrical spring liner, it was proposed to make the spring liner conical and the remaining surfaces of the parts mating with it cylindrical, in order to ensure tight fit on cylindrical surfaces [21].

Tribocoupling (Figure 10) is comprised of the shaft 1, the external ring 2 and the spiral box 3 in the form of coiled spring positioned between them. The spiral box is movable, conical with the cone angle from 1° to 5° , herewith, the spring wire diameter d equals to one half of the gap between the shaft diameter D and the diameter of box opening $D+2d$. Herewith, it is installed with allowance at edges as well as with allowance at internal and external surfaces for constant ratchet effect [22, 23].

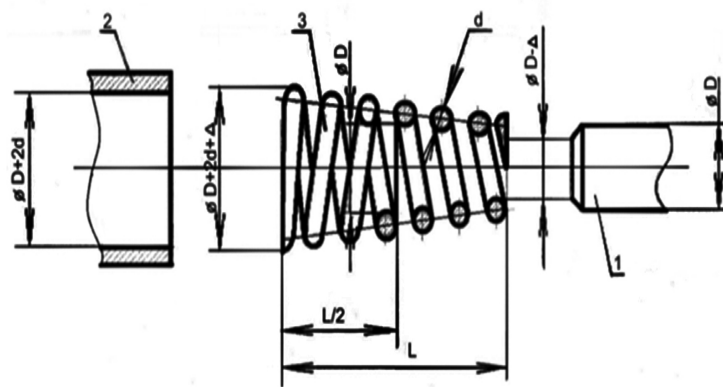


Figure 10: Sliding Bearing for Reciprocating/Rotary Motion.

In a conical spring insert, the necessary mating conditions are provided automatically by the design of the insert. The conical insert in the sliding bearing for the reverse-rotation movement has specific fit. So at rest, the conical spring liner, at half its length (from the side of the smaller diameter), has an interference fit that reduces to zero in the middle of the liner on the shaft and the same fit along the sleeve from the side of the larger diameter of the spring liner. Such landings provide a guaranteed interference fit on both working surfaces, which satisfies one of the conditions of bearing operability - the creation of plastic deformation on working surfaces [6]. It was developed a mathematical model of the process of functioning (resizing) of a conical movable spring insert of a sliding bearing of a shock absorber silent block [21].

The mathematical model is based on the mathematical model for a cylindrical spring insert, but since the conical spring insert differs from the cylindrical, several assumptions were made in the calculation.

When calculating the conical spring insert, we assume that it is made of a 65G spring wire of square cross section with a square side of 1.4 mm, this wire was chosen because, it is most suitable for manufacturing the spring bush of the silent block of the rear shock absorber of the VAZ family which is the object of study (Figure 11).

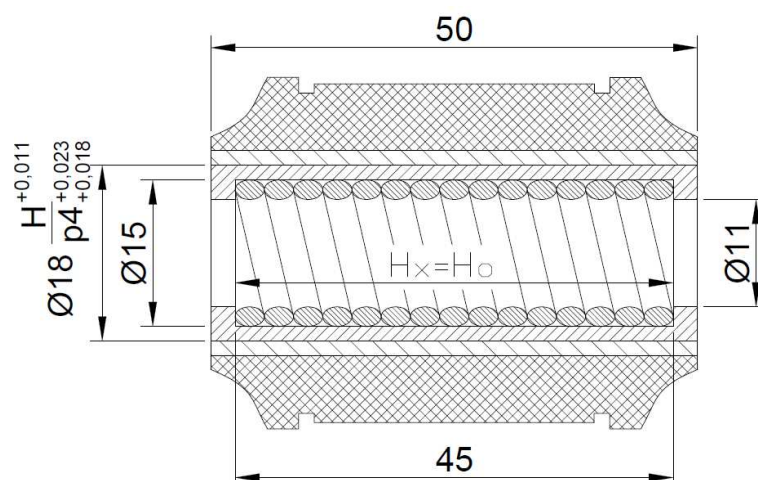


Figure 11: Schematic view of Damper Rubber Bushing with Movable Conical Box.

In the scheme, the spring insert has a cylindrical section (in the general case of calculation). At the same time, it can be ground to the required size, both inside and outside. When calculating the conical spring insert, the following assumptions are made:

The following assumptions are made:

- spring wire of steel, grade 65G;
- absolute linear deformation of spring box f_x is 1 mm;
- the length of spring box in loaded state H_x equals to H_0 because it is unloaded;
- the pitch angle of spring box in unloaded (free) state is $\alpha = 1.83^\circ$;
- the initial pitch angle of screw beam axis of unloaded spring box is $\alpha = \alpha_0$.

Taking into account that the conical spring box differs from cylindrical one, new variables have been applied:

r_1 and r_2 are the minimum and maximum average radii of coils of conical spring box;

D_{0min} is the average diameter of the minimum coil of conical spring box;

D_{0max} is the average diameter of the maximum coil of conical spring box.

Further predictions have been made for spring box in the form of stiff beam.

The bending stiffness is:

$$B = Ea^4 / 12, \quad (1)$$

Where $E = 20 \cdot 10^4 \text{ MPa}$ is the elasticity modulus of the first kind.

The torsion stiffness is:

$$C = \eta a^4 G = \eta a^4 E / 2 \cdot (1 + \mu), \quad (2)$$

Where μ is the Poisson ratio of parts.

The height (length) of spring box in unloaded state is:

$$l_0 = \pi D_0 / \cos \alpha_0, \quad (3)$$

The length of working portion of spring box is:

$$H_0 = l_0 \sin \alpha_0, \quad (4)$$

The axial force at which the conical spring box is compressed to the maximum is:

$$P_{kn} = \frac{4 \cos^2 \alpha_0}{D_0^2} \cdot \left[C(\sin \alpha - \sin \alpha_0) - B \sin \alpha \left(1 - \frac{\cos \alpha_0}{\cos \alpha} \right) \right], \quad (5)$$

The allowable bending moment is:

$$M = M_0 = - \frac{P D_0 (B - C) \sin 2\alpha_0}{4(B \sin^2 \alpha_0 + C \cos^2 \alpha_0)}, \quad (6)$$

The average spring hook is:

$$R = M / F, \quad (7)$$

The response force upon loading of conical spring box with constant step to the height H_x is:

$$F_x = \frac{f_x \cdot 2 \cdot C}{p \cdot n \cdot (r_2^2 + r_1^2) \cdot (r_2 + r_1)^2}, \quad (8)$$

where $f_x = H_0 - H_x$; $x=1,2,3$; r_1, r_2 are the minimum and maximum rated average radius of working portion of coils of conical spring box.

Compression under the action of force P of conical spring box is as follows:

$$\lambda \approx \frac{0,25(H_0 - H_m)}{(1-n)} \cdot \left(4 - 3\sqrt{\frac{P_{ini}}{P}} - \frac{P}{P_{ini}} n^4 \right), \quad (9)$$

where $(r_1 - r_2) \cdot i$; i is the number of working coils; P_{ini} is the initial foil of coil compression; H_m is the height of completely compressed conical spring box:

$$H_m = \sqrt{(ia)^2 - (r_2 - r_1)^2}, \quad (10)$$

$$n = \frac{r_1}{r_2}, \quad (11)$$

where n is the number of working coils.

The allowable size variation of conical movable spring box of bearing ΔD (with the radius $[f]$) in final form is as follows:

$$\Delta D(\Delta \lambda) = - \left[(P \Delta \lambda^3 \sin(\alpha_0)) \cdot \left(\frac{1}{2C} - \frac{\cos(2\alpha_0)}{4B \cos(\alpha_0)^2} \right) \right] - \quad (12)$$

$$- M \frac{\Delta \lambda^2}{2 \cos(\alpha_0)} \cdot \left(\frac{\sin(\alpha_0)^2 \cdot 2}{C} + \frac{\cos(2\alpha_0)}{B} \right)$$

$$[f] = \frac{3\pi[P]R^3}{Ea^4}, \quad (13)$$

Where $[P]$ is the allowable force for increase in radius of spring ring;

$$[P] = \frac{[\sigma]a^2}{\left\{ \frac{6R}{a} + 1 \right\}}, \quad (14)$$

where $[\sigma]$ is the allowable bending stress.

The calculation results showed that the proposed mathematical model for a conical spring insert is more accurate than the model for a cylindrical spring insert, because it has a more accurate correlation and an extended size range of use.

The developed mathematical model is used for theoretical calculations of the silent block bearing included in the shock absorber repair kit. The developed bearing is the main element of the innovative repair kit for the silent block of the car shock absorber (Figure 12).

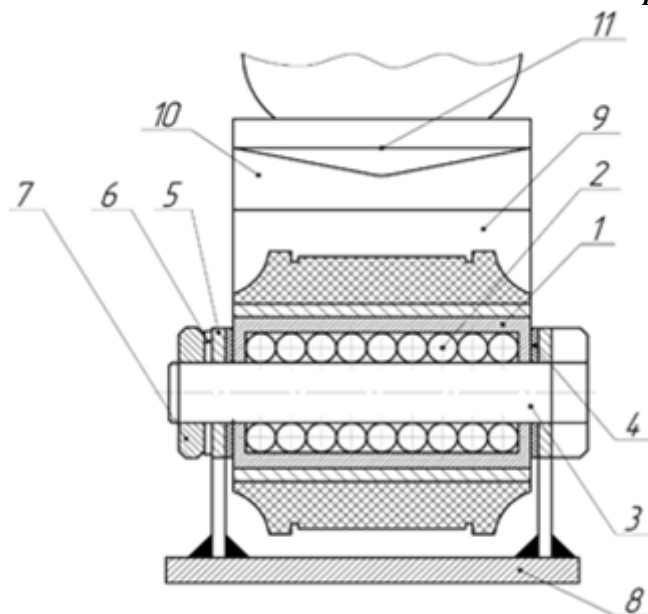


Figure 12: Installation Diagram of the Silent Block of the Shock Absorber with a Movable Conical Spring Insert for Cars of the VAZ Family.
(1-steel sleeve; 2-conical spring insert; 3-bolt; 4-rubber seal; 5-lug; 6-washer; 7-nut; 8-support; 9-silent block aluminum bush; 10-bushes silent block; 11- shock absorber lower housing)

3. RESULTS AND DISCUSSIONS

Theoretical assumptions for development of sliding bearing with spring boxes are proved by laboratory studies of experimental repair kit of damper rubber bushing of passenger car performed with high precision instruments, as well as in working environment [8]. Practical significance is that the developed repair kits of damper rubber bushing are characterized by operation lifetime higher by 1.5...2 times in comparison with regular kits operating in reciprocating/rotary mode.

Some results can be applied for development of new designs and increase in operation lifetime of existing agricultural equipment [13–15], for instance, in horticultural activators, such as one-, two-, and four row furrow shapers (BR-1, BR-2, BR-4) and console machine (BRK-1).

4. CONCLUSIONS

The obtained results can be applied for development of new engineering solutions, aimed at efficient operability maintenance of transport and agricultural equipment based on recovery improvement of couplings operating under high loads in reciprocating/rotary modes (for instance, steering, drive shafts, vehicle suspension) using repair kits.

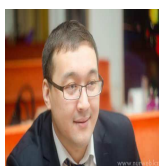
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